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A SIMULATED UNIT INJECTION SYSTEM

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SUMMARY

Rate-of-discharge curves that show the discharge characteristics of an injection system having a very short fuel passage are presented. The rate of discharge closely follows the rate of displacement of the injection-pump plunger for open nozzles in which the maximum calculated pressures at the orifice do not exceed a certain value, which is dependent on the particular injection pump. With small orifices and high pump speeds, the rate of discharge does not follow the rate of plunger displacement because the higher discharge pressures result in increased leakage past the pump plunger with a corresponding decrease in discharge rate and in fuel quantity. The rate of discharge is not directly related to the rate of plunger displacement with automatic injection valves having closed nozzles. The types of pump check valve tested did not control the rate of cut-off or the discharge rate but they did affect the injection lag. Increase in valve opening pressure slightly increased the injection lag. Use of the short fuel passage eliminated the formation of secondary discharges.

INTRODUCTION

In the adaptation of a fuel-injection system to a high-speed internal-combustion engine, injection characteristics peculiar to the individual injection system develop. This condition necessitates testing the injection system under all operating conditions.

Experiments in this laboratory (references 1 to 4) with a cam-operated injection pump have shown that secondary discharges occur above certain pump speeds and that, for various combinations of orifice diameter, injection-tube diameter, injection-valve opening pressure, and pump speed, fluctuating discharges appear. These characteristics are explained by the variation in pressure at the orifice caused by the pressure-wave phenomenon existing in the injection tube during the injection period.

Shortening the injection tube results in a decrease in the time of a pressure-wave cycle and a reduction in the friction loss, the injection lag, and the volume of oil in the system. Elimination of the injection tube should reduce the variation in the build-up of pressure at the pump and at the orifice. With the orifice placed directly at the pump chamber, the pressure build-up at the orifice should be directly controlled by the cam outline. The ideal condition is that the discharge at the orifice should vary directly with the plunger displacement, thus simplifying control of the injection process.

An injection system incorporating the injection valve and pump plunger with practical elimination of the injection tube is termed a "unit injector." Rate-of-discharge tests were made with such an injection system on which the injection passage was made as short as 3 inches. The injection characteristics were determined for changes in pump speed, orifice diameter, cam outline, injection-valve design, and type of pump check valve.

APPARATUS AND METHODS

The apparatus used was that described in reference 1 with the pump so mounted that the injection tube was as short as practicable.

A single-cylinder Bosch pump having a 10 mm diameter plunger was employed throughout the tests. Three cam outlines were used. The pump characteristics with the three cams are shown in figure 1. Cam 1 has a high initial velocity at port closing and maintains an increasing velocity for 15° during the injection period. Cams 2 and 3 are very similar and were constructed with a slow initial plunger displacement so that the initial rate would not be affected by the rate of port closing. Two positions of port closing were used as indicated at a and b on the plunger-lift curves. The data shown in figures 3 to 9 were obtained with port closing at a and those in figure 10 with port closing at b. The port closing for cam 1 on the rate-of-discharge curves is indicated at zero pump degrees and the port opening is indicated by the vertical line close to the cut-off. These positions were obtained statically. The values under the curves indicate the fuel quantity ($\times 10^{-5}$ pound) discharged per injection.

The two injection valves tested (fig. 2) represented possible types for the construction of a unit injection system. The simplest type (valve 1) was a ball check in combination with an open nozzle having orifice diameters of 0.010, 0.020, 0.028, and 0.040 inch. The valve opening pressure could be varied up to 3,600 pounds per square inch. Valve 2, a differential-area, lapped-stem injection valve, was tested with orifice diameters of 0.022, 0.033, and 0.040 inch at a valve opening pressure of 3,500 pounds per square inch.

The following conditions were maintained constant unless otherwise specified:

Pump speed	1,000 r.p.m.
Valve opening pressure:	
Valve 1	400 lb./sq. in.
Valve 2	3,500 lb./sq. in.
Orifice diameter:	
Valve 1	0.028 in.
Valve 2	0.033 in.
Fuel-passage length (plunger to orifice):	
Valve 1	3-1/4 in.
Valve 2	8-1/4 in.
Injection-tube inside diameter . .	0.125 in.
Fuel pressure at inlet to pump . .	50 lb./sq. in.
Fuel specific gravity.	0.85
Pump check valve	Bosch

RESULTS AND DISCUSSION

The variation in the rate of discharge with cam outline for valve 1 (fig. 3) is definite. The rate of discharge agrees closely with the rate of plunger displacement during the increasing-velocity portion of the cam. The rate of discharge, however, does not reach the same maximum value nor fall so rapidly as the rate of plunger displacement during the decreasing-velocity portion of the cam. For valve 2, variation in cam outline does not change the rate of discharge during the initial period and the variation during the whole period is not directly related to the plunger displacement. With valve 1, control over the injection process is evidently exercised at the pump and, with valve 2, the control is fundamentally at the injection valve.

The rate-of-discharge curves taken at various pump speeds and with various orifice diameters are shown in figure 4. Rate curves could not be obtained for the smaller orifices at high pump speeds because of shock loading on the driving shaft of the bracket assembly. Injection began earlier as the pump speed was increased or the orifice diameter decreased; it also occurred before the static timing of port closing.

A plot of the pressure calculated from the maximum rate of discharge (reference 5) indicates the increase in pressure obtained with increase in pump speed or decrease in orifice diameter (fig. 5). To this increase in pressure and subsequent increased leakage at the pump plunger is attributed the failure of the rate of discharge to follow the plunger displacement for all orifice diameters and pump speeds.

Pressures calculated from the rate-of-discharge curve at a point 6° from cut-off at 1,000 r.p.m. are 6,100, 3,000, and 1,100 pounds per square inch for the 0.020-, 0.028-, and 0.040-inch orifices, respectively.

The lapped length of plunger covering the cut-off ports is about 0.050 inch. The clearance of the lap at this point is high, owing to constant use of the pump over a period of several years. On the assumption that the mean clearance is 0.0005 inch, the leakages are 0.0000134, 0.0000048, and 0.0000015 pound per degree (table I). The summation of the actual discharge rate with this assumed leakage rate is 0.0000302, 0.0000277, and 0.0000287 pound per degree, indicating a close agreement with the actual rate of fuel displacement. To overcome this leakage factor to some extent would require an increased length of lap sealing the ports and a greater sleeve diameter. The increase in mean clearance due to the pressure increase approaches in value the initial mean clearance.

TABLE I

Calculated Leakage Due to Pressure

Orifice diameter (in.)	Calculated pressure (lb./sq.in.)	Increase in mean clearance due to pressure increase (in.)	Calculated leakage (lb./deg.)	Discharge rate (lb./deg.)	Summation of leakage and discharge rate (lb./deg.)	Rate of displacement (lb./deg.)
0.020	6,100	0.00025	13.4×10^{-6}	16.8×10^{-6}	30.2×10^{-6}	29.5×10^{-6}
.028	3,000	.00012	4.8	22.9	27.7	29.5
.040	1,100	.00005	1.5	27.2	28.7	29.5

The type of pump check valve tested has little effect on the time or rate of cut-off (fig. 6). It does, however, change the beginning of injection. The start of injection was later with the lapped shoulder check valve (Bosch check valve). The opposite effect was obtained with a differential-area valve in previous tests in which longer injection tubes were used (reference 1). The increased lag of the Bosch check valve is due to the displacement of the lapped shoulder.

Volume of lapped displacement = 0.0026 cubic inch.

Lift of plunger = 0.0026/0.122 = 0.0210 inch.

The plunger motion indicates that this lift occurs in 3 pump degrees, which checks the difference shown in figure 6. With the longer injection tubes of the previous tests, the displacement of the lapped shoulder was insufficient to reduce the pressure in the injection tube to a low value so that the injection lag was increased with removal of the check valve.

The results obtained for both the poppet check valve and no check valve were similar indicating that the pressures in the injection system with the two valve arrangements were the same. For these conditions, the residual pressure in the injection tube must be lowered to the pump inlet pressure.

An increase in the valve opening pressure of valve 1 (fig. 7) slightly increased the injection lag. This effect is the same as obtained in previous tests with differential-area valves. When valve 1 was tested connected to a high-pressure hand pump, the fuel flow appeared restricted at the higher valve opening pressures. Normally this type of valve is used with light spring loading and an open nozzle, and in these tests restriction to flow did not occur.

The variation in the injection rate with valves 1 and 2 is striking. For the same rates of fuel displacement, as shown in figures 3 and 8, the discharge rates are distinct and characteristic. For varying pump speed, there is less change in the rate with valve 1 than with valve 2. Valve 2 at 500 r.p.m. shows a two-lobe injection rate obtained under certain conditions of operation (reference 2). The two valves are normally used with a low valve opening pressure for valve 1 and a high valve opening pressure for valve 2, but the general characteristics are unaffected by the valve opening pressure. Valve 1 has primarily a reducing action in that the pressure at the orifice must be less than the pressure at the entrance to the check valve. The hydraulic pressure tends to balance with the ball open and with the differential pressure action obtained by the spring load and the flow restriction. Reflection of the pressure wave occurs at the ball and seat and at the orifice.

Valve 2, with its differential-area stem, opens quickly when the valve opening pressure is reached and the pressure build-up at the orifice is not necessarily restricted. Reflection of the pressure waves occurs at the stem seat, which is normally close to the orifice. With large orifice diameters, fluctuations in the discharge may occur, owing to the drop in pressure at the orifice from discharge and to the sudden increase in pressure due to the arrival of a pressure wave at the orifice.

Figure 9 shows that valve 1 has an increasing initial rate of discharge, which is maintained with increase in fuel quantity. Valve 2 has an increasing initial rate of discharge, twice that of valve 1, and reaches a maximum rate above which it does not increase with increase in fuel quantity. As a result, valve 1 has a peaked rate curve and valve 2 a flat-top rate curve. The superior control of the open nozzle is evident from these curves. The injection period and the maximum rate of the two valves are comparable at the higher throttle settings. The closed nozzle has a high rate of discharge at low throttle settings, which results in a shorter injection period than the open nozzle.

In order to check the effect of the slow closing of the ports with cam 3, a 0.030-inch spacer was placed under the plunger sleeve to delay the closing of the ports (fig. 1) until a period of higher plunger velocity. The initial rate of discharge was increased with valve 1 and little affected with valve 2 (fig. 10). In order to obtain about the same fuel quantity as in the previous tests, a greater portion of the decreasing-velocity portion of the cam was used. The rate of discharge of valve 1 also agreed very closely with the rate of fuel displacement during this portion of the cam outline.

CONCLUSIONS

The following conclusions are based on tests with a fuel-injection system using a 3-inch fuel passage and an open nozzle.

1. The rate of discharge can be directly controlled by the rate of plunger displacement under operating conditions in which the calculated fuel pressure at the orifice does not exceed a certain maximum value, which for the system tested was about 3,000 pounds per square inch.

2. The rate of discharge is not independent of orifice diameter and pump speed when high fuel pressures are reached because of increased fuel leakage at the pump control ports.

3. Secondary discharges are eliminated as compared with a system having a long injection tube.

4. The rate of discharge is affected by the rate of of closing of the pump control ports.

The following conclusions are based on injection-valve design:

1. With open nozzles or with a ball check between the pump plunger and the orifice, the injection rate was controlled at the pump.

2. With closed nozzles having a differential-area lapped stem, the injection rate was not controlled at the injection pump.

3. An increasing initial rate of discharge is maintained with increase in fuel quantity up to the point of cut-off for the open nozzle.

4. The closed nozzles reach a maximum rate of discharge at low throttle settings; the rate does not increase with increase in fuel quantity, resulting in a flat-top rate-of-discharge curve.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., September 13, 1938.

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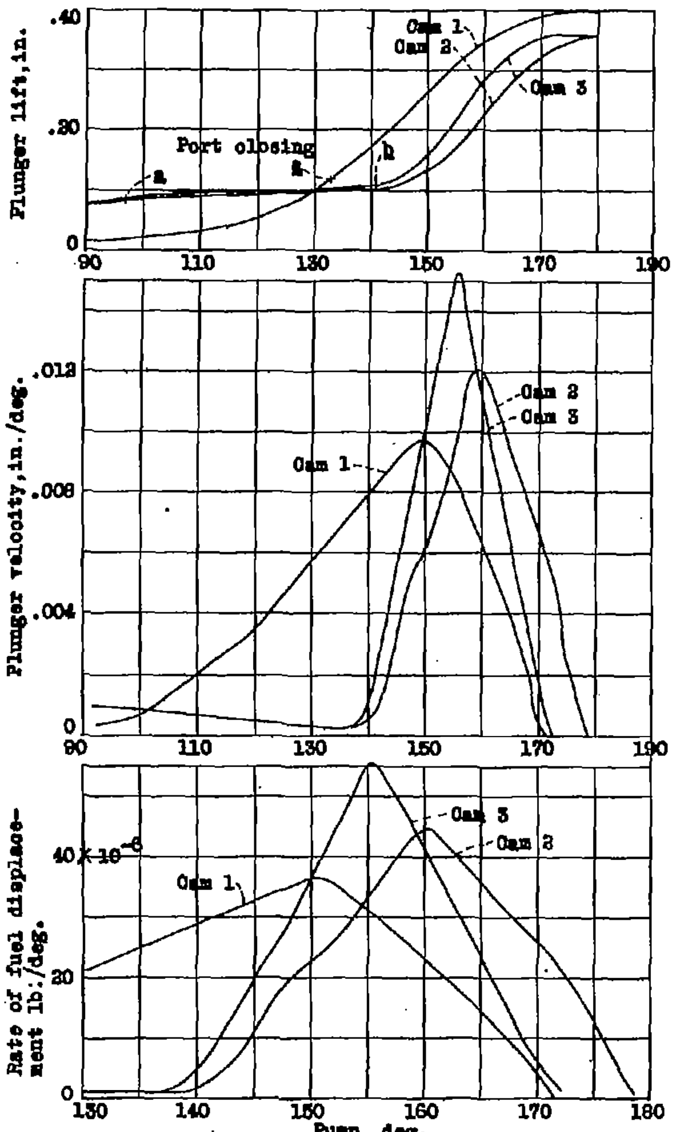
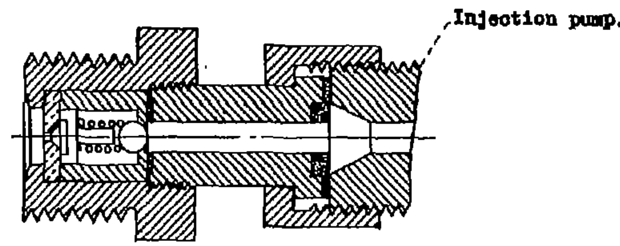


Figure 1.- Pump characteristics.



Valve 1.

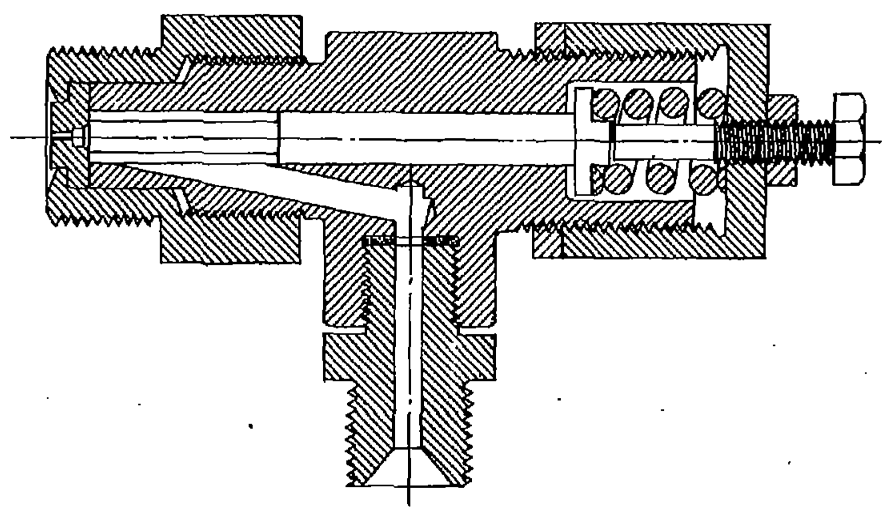

Valve 2.

Figure 2.- Injection-valve assemblies.

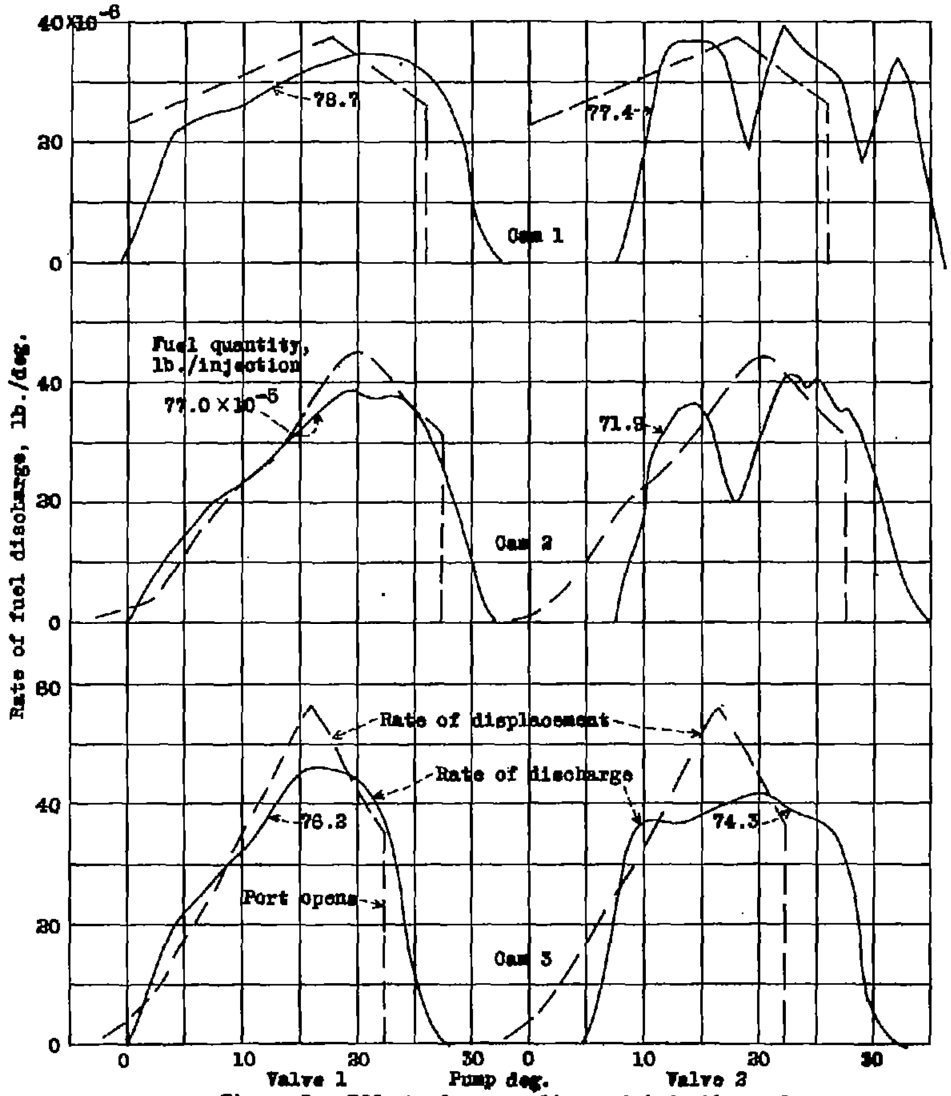


Figure 3.- Effect of cam outline and injection-valve design on the rate of discharge.

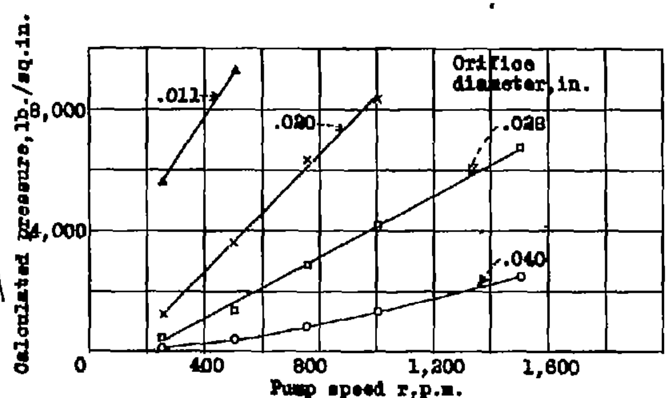


Figure 5.- Calculated pressure at the maximum rate of discharge. Valve 1; cam 1.

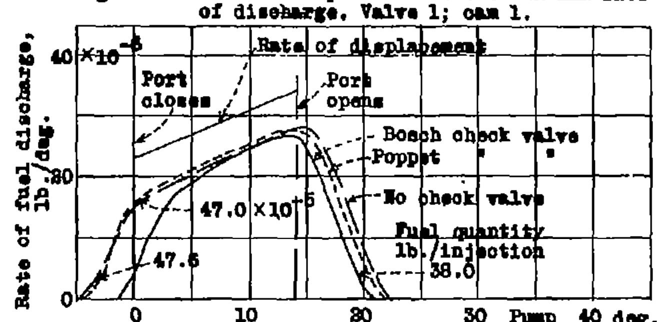


Figure 6.- Effect of type of pump check valve. Valve 1; cam 1.

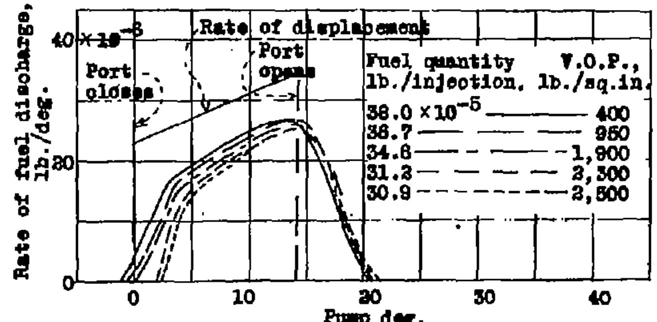


Figure 7.- Effect of injection-valve opening pressure. Valve 1; cam 1.

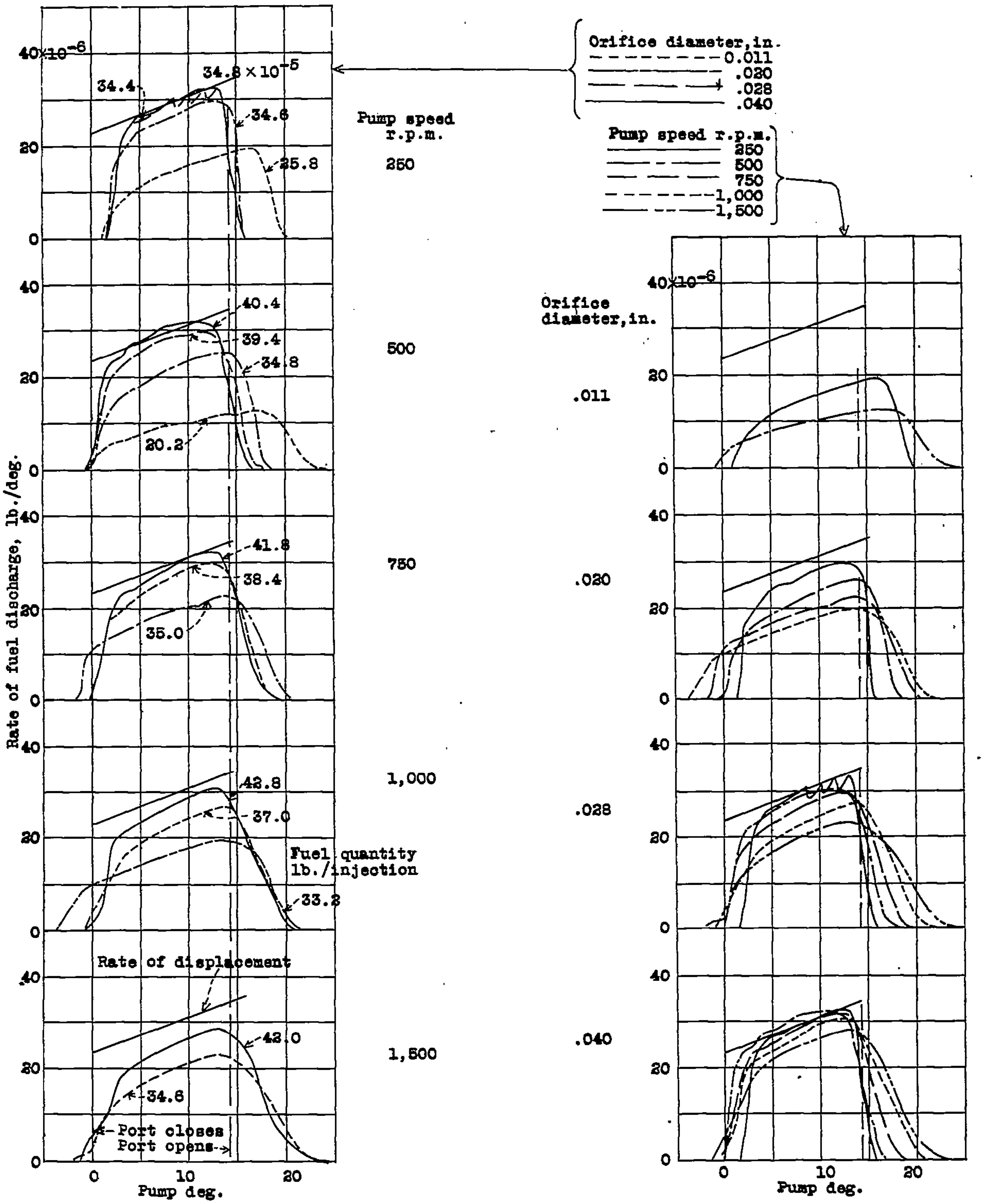


Figure 4.- Effect of pump speed and orifice diameter on the rate of discharge. Valve 1; cam 1.

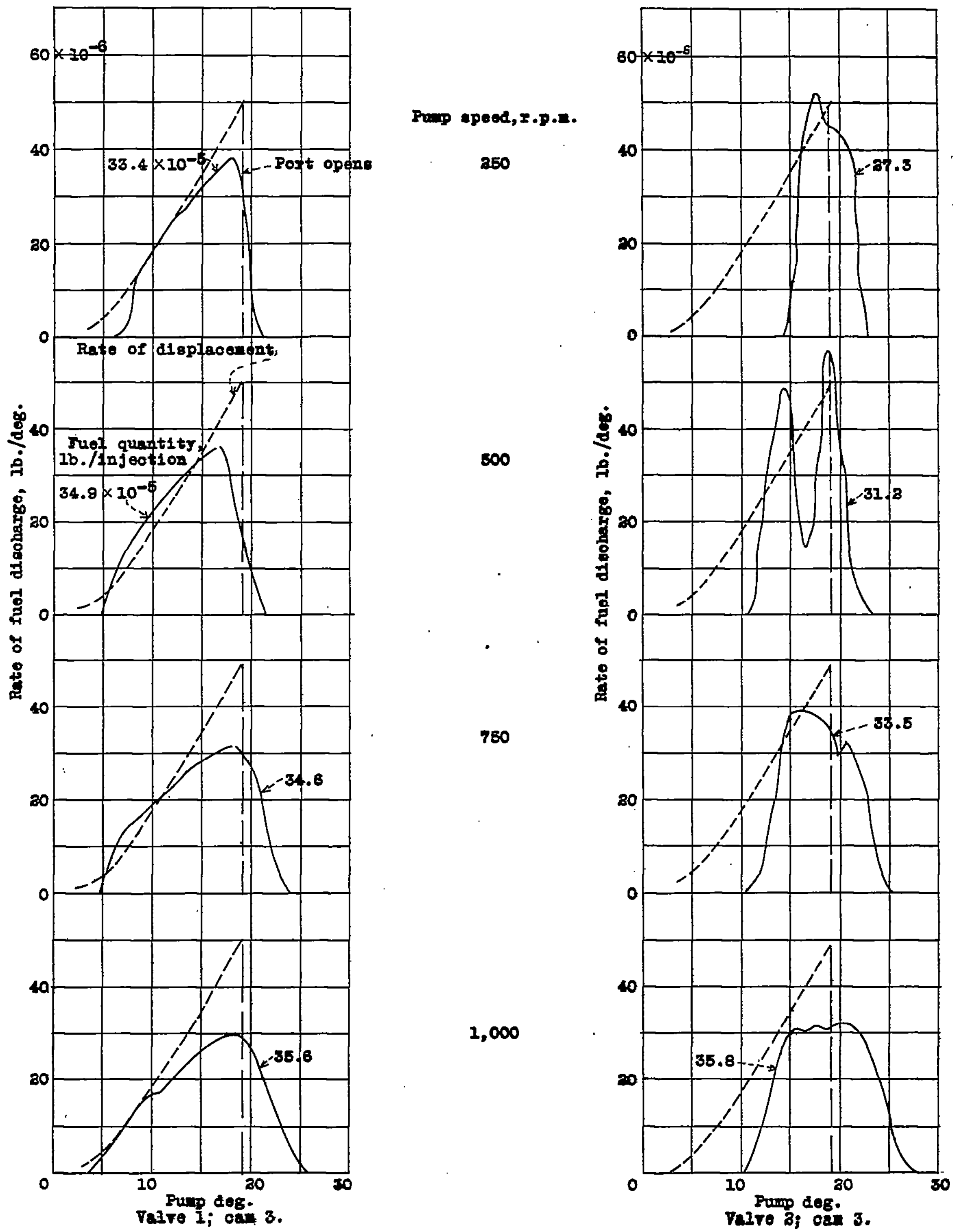


Figure 8.- Effect of pump speed and injection-valve design on the rate of discharge.

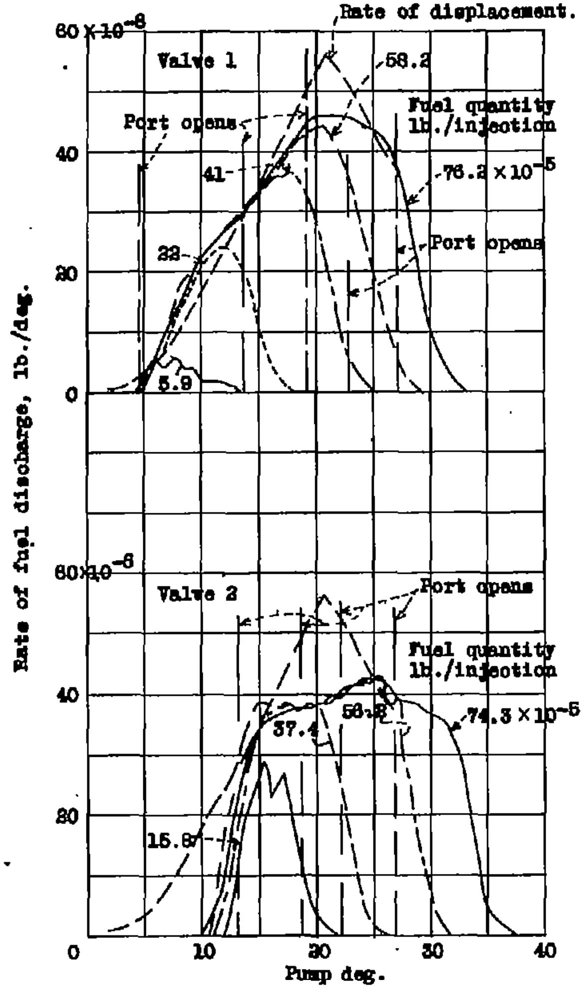


Figure 9.- Effect of fuel quantity and injection-valve design on the rate of discharge. Port closing, a; orifice diameter, 0.040 inch; cam 3.

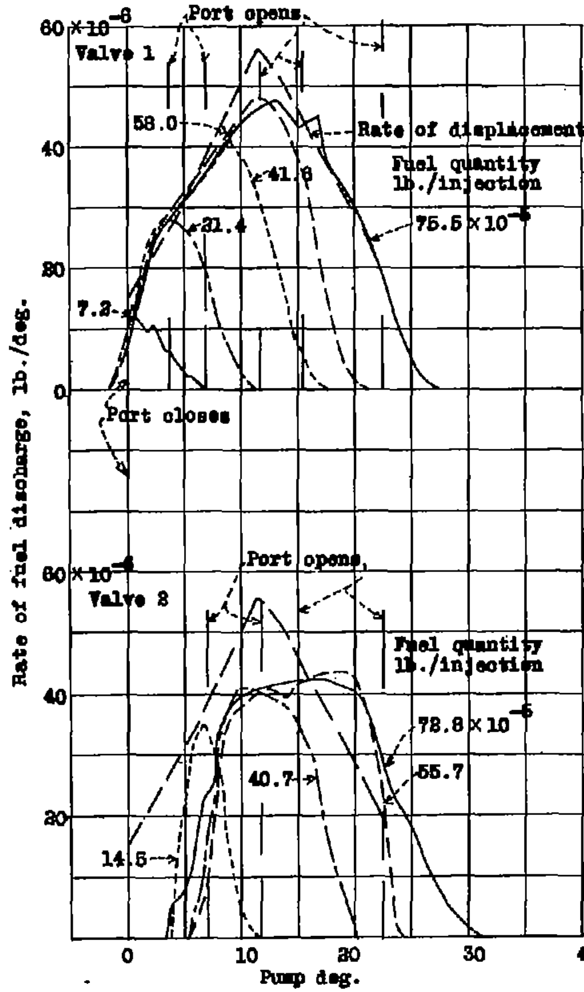


Figure 10.- Effect of fuel quantity and injection-valve design on the rate of discharge. Port closing, b; orifice diameter; 0.040 inch; cam 3.